FRICTION TESTS OF LUBRICATING OILS

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1913

FRICTION TESTS OF LUBRICATING OILS —INFLUENCE OF RATE OF FLOW

A THESIS

PRESENTED BY

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TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

HAVING COMPLETED THE PRESCRIBED COURSE OF STUDY IN

MECHANICAL ENGINEERING

MAY 29, 1913

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FRICTION TESTS OF LUBRICATING OILS
-INFLUENCE OF RATE OF FLOW-

With the view of determining the influence of the different rates of flow of oil on the bearing friction, the following series of comparative tests, ranging from the low restricted rates of feed to the "flooded" or oil bath method, were carried out on a railroad lubricant testing machine. This machine was selected for the tests because of the wide variation of bearing pressures, and of the promtitude of adoption, which could be obtained by its use. Then too, the condition of constant speed was afforded by the variable speed motor used to drive this machine.

The most serious loss encountered in the manufacturing world is that of the waste power caused by friction. Prof. Peabody states that 5 per cent and often greater than 15 per cent of the indicated horse power of a steam engine is expended in overcoming the frictional resistance; whereas Archbutt says that from 40 per cent to 80 per cent of the 10,000,000 h.p. used in Great Britain is consumed by friction;

probably in the United States this fraction would be much greater due to the cheapness of fuel. Much of this needless waste caused by friction as well as that due to neglect in reclaiming the oil is gradually being reduced to a minimum by the installation of oiling systems.

The lubrication of bearings, guides, and all external rubbing surfaces may be performed in a number of ways. These parts may be given an intermittent application of oil; they may also be supplied restricted rates of feed; or they may be flooded with oil. The intermittent feed, which is effected by the occasional use of an oil can, is mostly limited to moving parts carrying light pressures which do not easily permit the use of other systems. The restricted feed, or method of lubrication by means of cups from which oil is fed to the rubbing surface by drops is in majority in the average plant. The flooded arrangement is effected by allowing a continuous flow of oil, which is forced to the various parts either by gravity or pressure from a pump, to completely "flood" the bearing. In the latter system the oil is used over and over again, that lost by leakage and depreciation being replenished by the addition of new oil.

As was mentioned before, the machine employed in the series of tests was a railroad-lubricant testing machine. This machine, as shown in the enclosed drawing, consists of a shaft, which has a pulley, carried between two bearings, and driven by a belt from an overhead line shaft. The shaft of the machine extends beyond one of the bearings so that on the overhanging part is a pendulum which contains the test brasses. To insure that all of the surface does actually rub against the shaft, the brasses were so cut that instead of the width of the projected area being that of the diameter of the shaft, it was approximately three-quarters of the diameter. To these brasses a pressure is exerted by two heavy springs placed one inside the other. When these springs are compressed between the lower end of the pendulum, which is a pipe fitted with a castiron plug, and a nut on the upper part of the springs, the reaction on the plug is transferred through the bolt to a plunger that is pushed against the lower journal bearing. The magnitude of the total pressure on the bearing is indicated by a pointer attached to the nut, which moves along a graduated scale on the pendulum. The graduations were laid out according to the manu-

facturer's calibration curve for these springs.The deviation of the pendulum is measured by a graduated arc fastened to the frame of the machine and a pointer which is attached to the upper part of the pendulum. The bearing temperature is given by a thermometer which is placed in a hole which is drilled through the bearing into the babbitt metal. This hole is filled with oil so that the bulb of the thermometer may be completely submerged.

To make the machine more sensitive, that is, to increase the deflection for the same amount of friction, a large cast iron ball was added above the bearing that the moment due to the necessary heavy construction of the pendulum might be reduced. In case the friction becomes too excessive, the counterbalance can be lowered on its rod and securely held in position by means of a set screw. To prevent the pendulum from doing injury to anyone who might be in its path, a guard was placed on the base of the machine so as to restrict the pendulum to an arc of 25 degrees from its vertical position, when an excessive frictional load is applied.

The feeding apparatus used to regulate the supply of oil to the test bearing is that of the gravity

type. The oil, which is contained in a tank mounted on a platform scale, which in turn rests on a movable suspended shelf, is supplied to the bearings through flexible rubber tubing and branches of piping leading to the brasses. Inserted in this piping are two small stop cocks which are used to control the rate of flow. From the brasses the oil is distributed over the journal through grooves run diagonally across the babbitt face from the inlet holes, thus giving even and equal distribution. These channels are carefully gaged for an even flow to prevent dry spots or streaks appearing on the journal accompanied by sudden greatly increased friction. The grooves which are 1/8 inch wide and 1/16 inch deep carried the oil on the top surface lengthwise of the bearing so that the moving surface passing the grooves or chamfers was bathed in, and coated with oil. In the lower bearing the grooves were cut so as to wipe off the oil, which has squeezed over toward the end of the bearing, and thus prevented as little as possible working out. That which did pass out was drained down through holes in the plunger, nut, and plug to a receptacle where it was caught.

After enough oil had gathered in the reservoir, the oil was passed through a "White Star" filter where

the anti-lubricating matter was taken out so as to render it fit to be used again. This reuse of the oil did not change the characteristics to any appreciable amount as far as could be judged from the results obtained. For, data acquired on different runs fulfilling the same conditions excepting that in one case new oil was used and in the other, oil several times reclaimed, showed practically no discrepancies. According to tests made on a Martin's oil testing machine of oil used over and over again and recovered under proper conditions, the characteristics showed no variation between the values before and after use outside the limits of possible observation. But use slightly increases the density, although it does not materially change the viscosity.

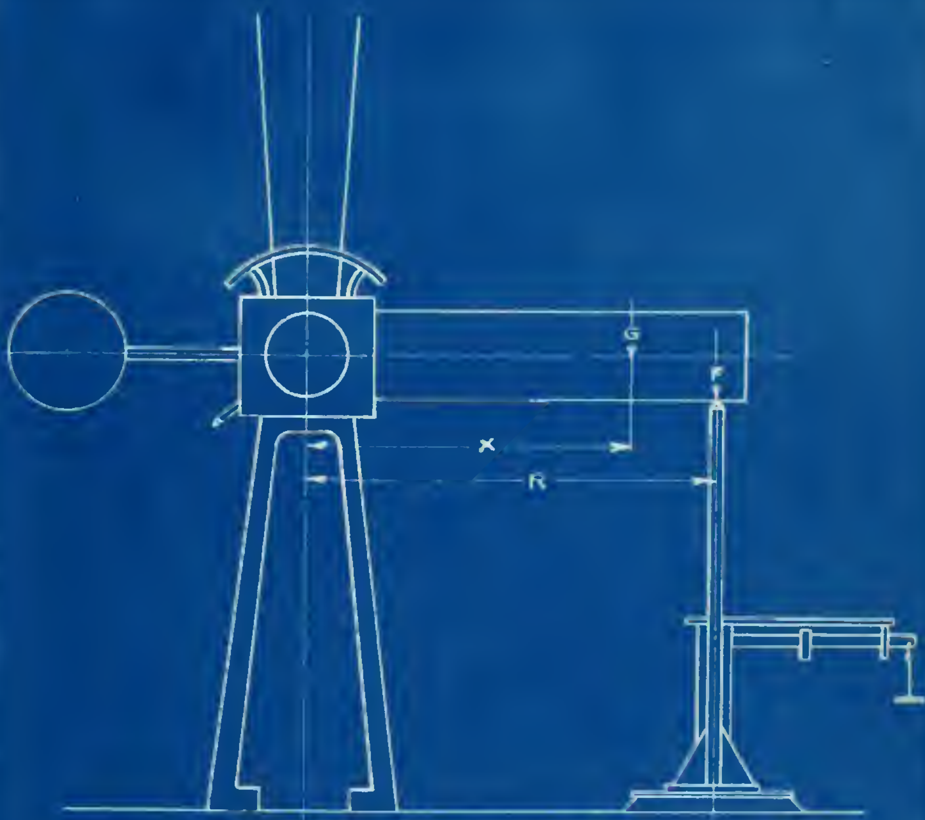
Before the machine was used for the tests, it was entirely taken apart so that it could be thoroughly cleaned with gasoline in order to remove dirt, grit, and traces of other oils previously used.

While the machine was in this condition the various parts were placed on a platform scale and the entire mass weighed.

Although the machine had been used, the moment had never been actually determined. To ascertain

this the pendulum, with brasses removed, was supported in a horizontal position on a three-cornered file as follows: with the pendulum hanging vertically a line was drawn on the bearing box at the intersection of the horizontal plane thru the journal axis; the pendulum was then swung into a horizontal position and the line of contact with the file was made to coincide with this line just found. A strut upon a platform scale supported the free end of the pendulum, which was then leveled by means of wedges and a spirit level. The weight indicated by the platform scale was recorded as well as the distance between the centers of support. Of course from the total weight, the weight of the strut and wedges had to be deducted. The accepted moment was taken as the average of the three sets of readings, the product of the net weight and the distance between the supports being the only calculation needed in each case. The omission of the brasses had no effect on the result because both of them weighed the same and occupied similar positions on either side of the axis so that even if they had been in place, each would have had a neutralizing effect on the other.

The determination of this moment is very important for upon its accuracy hinges the value of the



Arrangement of Apparatus in Determining Moment

$FR = GX$

(G = Force acting at c'tr. of gr.)

Arrangement of Apparatus during Test.
$Qr = G\ X \sin\theta$
(Q = Total Friction.)

tests, since it enters as an important factor into every calculation. With a knowledge of the weight of the heavy ball and a measurement of the distance moved from the original position a new moment may be calculated by adding or subtracting the product of the weight and distance, depending upon whether the ball is lowered or raised.

The moment of the machine having been determined, it was assembled into running order and, with the pendulum in a true vertical position, the pointer for indicating the deflection was set at zero on the graduated arc. So with everything calibrated, assembled, and with only the weight of the machine on the upper brass, power was thrown on, as well as a liberal feed of oil started.

After the machine had been running for several hours, conditions of temperature and speed were found to be constant so that the next change to be made was to regulate the flow of oil to a minimum. This was done by making several preliminary runs of fifteen minutes each until the desired feed was obtained, which was that of nearly approaching intermittent oiling. This method of regulation was necessary because of the fact that the valves and variance of the head had not

been calibrated as it was found impracticable due to the wide change of rates of flow caused by the different bearing pressures that were to be used. Had this been attempted, the regulation would have to have been calibrated for each bearing pressure.

When the desired regulation of feed was obtained, the first run of the series of friction tests was started by taking readings of the following:

Time.

Weight of oil and reservoir on scale.

Deflection of pendulum from normal position.

R.P.M. of shaft (kept constant)

Bearing temperature.

Room temperature.

These readings were recorded at intervals of ten minutes throughout the run, the duration of which was usually one hour. In a few cases where all conditions remained constant, and the rate of flow was sufficiently large so that small errors in scale readings might be neglected, runs one half hour in length were made, readings in such instances being taken every five minutes.

At the expiration of this run the flow was increased until a noticeable decrease of deflection

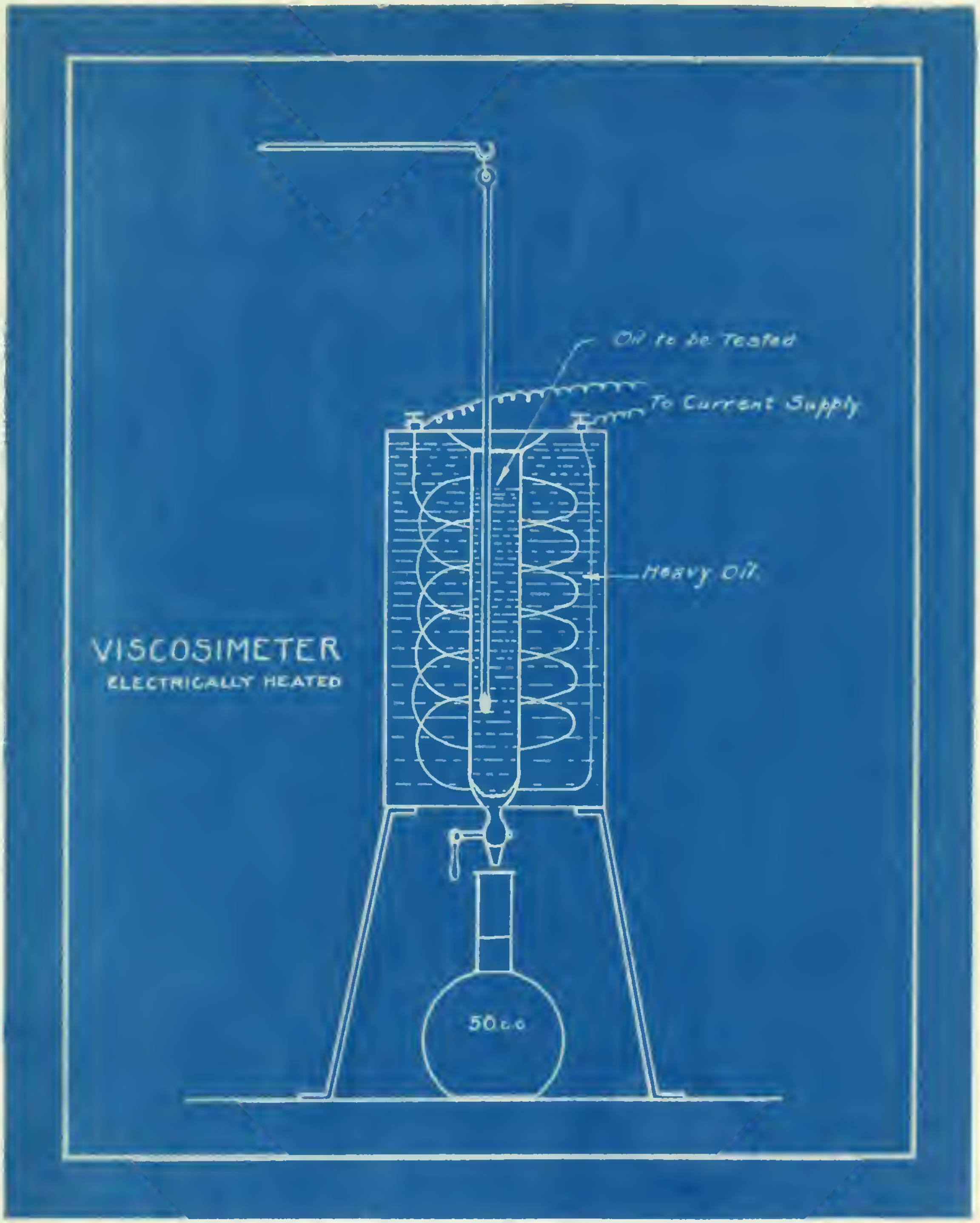
from the previous deviation was had. Then the machine was allowed to acquire constant bearing temperature, after which the second set of readings at ten minute intervals were taken. To get adequate data for each bearing pressure at least six of these runs had to be performed.

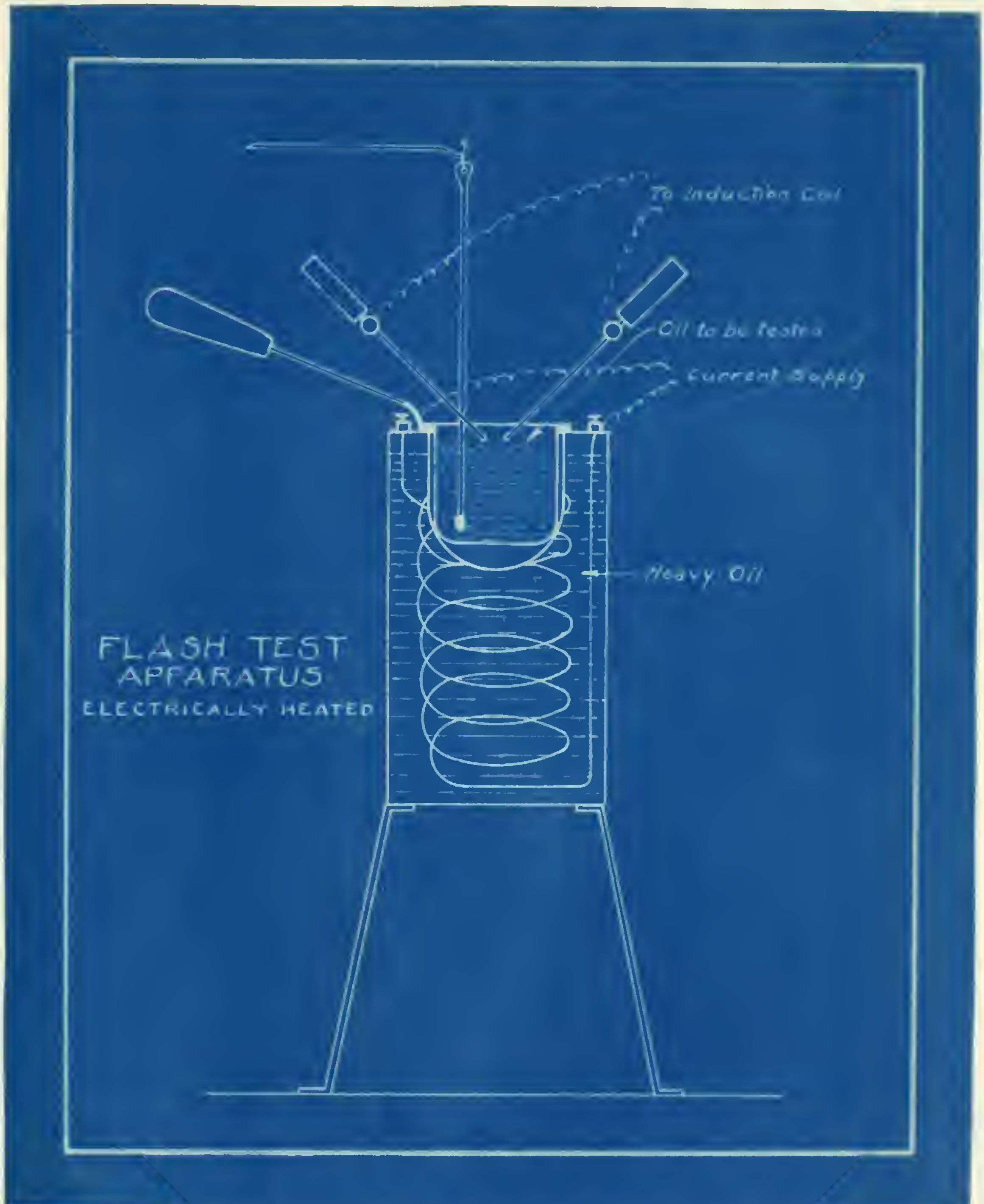
By deflecting the pendulum and using a wrench on the head of the bolt at the bottom, the pressure on the brasses was increased gradually until the pointer indicated a pressure of 1,000 lbs. This increase of bearing pressure had to be accompanied with a delay of readings because the bearing temperature for this frictional load was also increased. So during this intermission the flow was decreased to one as near the minimum as possible. When conditions were finally fulfilled another set of six runs of an hour duration were made.

This method of procedure was carried out up to a bearing pressure of 5,000 lbs. at which load it was found impossible to obtain reliable data. Then too at this pressure it was found impossible to get minimum rates of flow because of driving difficulties encountered such as slipping and throwing off of the belt. This latter difficulty necessitated

the complete throwing off of bearing pressure in order to set the journal in motion again.

The preceding series of readings of five different bearing pressures was again repeated for another sample of oil which not only differed from the first in body and density but in viscosity. Although both were mineral oils they were of different crudes, the first sample being from the eastern wells, and the latter from the southern fields. The physical properties of these oils were obtained by the use of the proper instruments. The viscosimeter used was the standard Tagliabue's instrument, the principle of which is shown in the accompanying sketch. The oil was raised to a temperature of 150°F., and the time for 50 c.c. to flow through the nozzle was recorded. These results, of course, have no value except comparatively, to obtain the relative viscosities of the two oils. The flash and fire points were obtained by means of a flash test machine of design similar to the one herein shown. The oil in the cup is heated and a spark from an induction coil passed across the surface at frequent intervals. The temperature at which the vapor rising from the surface is seen to flash is known as the flash point.


Oil to be Tested
To Current Supply
Heavy Oil.
VISCOSIMETER
ELECTRICALLY HEATED
50cc


To Induction Coil
Oil to be tested
Current Supply
Heavy Oil
FLASH TEST
APPARATUS
ELECTRICALLY HEATED

The heating is continued until the oil finally takes fire and burns, the temperature at which this occurs being known as the fire point. In obtaining the chill point a small quantity of the oil is placed in the bottom of a test tube which is cooled by an ice-salt mixture, until a thermometer inserted into the tube picks up lumps of the congealed oil on the end. The thermometer is held at an angle of 45° and the temperature at which the oil is seen to drop off is the chill point.

The theory of the machine is as follows:-Let

R = length of moment arm in inches

F = net weight on scale in lbs.

r = radius of journal in inches.

b = width of projected area of brass.

a = total projected area.

l = length of projected area of brass.

W = weight of pendulum complete.

P = total pressure on journal.

p = pressure per sq. in. of projected area.

T = tension on spring (read from graduation)

e = angle of deflection.

f = coefficient of friction

Q = total friction.

N = revolutions per minute.

From the construction of the machine it is readily seen that the pressure on the journal is made up of equal pressures caused by the action of the spring on the upper and lower brasses, and of the pressure due to the weight of the pendulum, which acts only on the upper brass. Since in the machine both brasses are loaded, the projected area is,

$$a = 2bl$$

But the total pressure is,

$$P = 2T + W$$

Then the pressure per square inch is,

$$p = P/2bl = \frac{2T + W}{2bl}.$$

Since the moment of friction is equal to the external moment of forces acting,

When $\theta = 90°$

$FR = Qr$ and is a maximum.

When at any other angle the moment arm = $R\sin.\theta$; and the moment,

$$FR \sin.\theta = Qr$$

From which, $Q = \dfrac{FR \sin.\theta}{r}$.

By definition, the coefficient of friction equals,

$$f = Q/P = \frac{FR \sin.\theta}{Pr}$$

But $P = 2T + W$

Therefore $f = \frac{FR \sin.\theta}{(2T + W)r}$.

It will be readily seen that $\frac{FR}{r}$ is constant for each individual machine, and the value is known as the "constant of the machine".

Thus,

$$f = \frac{C \sin.\theta}{P}$$

Since the principle is similar to that of the prony brake, the horse power loss is,

$$H.P. = \frac{2\pi NFR \sin.\theta}{12 \times 33000} = .00001586NCr \sin.\theta$$

SAMPLE 1.

Physical Characteristics.

Color	Reddish brown
Baumé (Degrees)	30
Specific Gravity	.8750
Pounds Per Gallon	7.29
Flash Point	417°F
Fire "	464°F
Chill "	32°F
Viscosity (referred to distilled water at 150°F)	1.77

S A M P L E 1 . BEARING PRESSURE

Total------------[illegible]70 lbs.
Per Sq. In. of Projected Area---14 lbs.

Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.00	0.000	6.60	.1149	.1410	177°F	73°	198	.664
.16	.165	2.95	.0514	.0632	134	72	200	.300
.33	.360	3.50	.0610	.0751	133	73	200	.355
.55	.605	2.93	.0510	.0627	129	74	197	.296
.59	.664	2.3	.0401	.0494	126	71	199	.258
.62	.680	2.45	.0428	.0525	133	72	201	[illegible]
.89	.980	1.94	.0340	.0419	117	72	202	.200
.95	1.050	2.02	.0352	[illegible]	125	73	203	.206

S A M P L E I. [illegible]

[illegible] 1 ---------- [illegible]
[illegible]
[illegible] ------ [illegible]

Rate of Flow		Deflection	[illegible]	[illegible] of	Bearing	[illegible]	[illegible]	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	[illegible]	Friction	Temp.	[illegible]		Loss
0.[illegible]	.010	7.00	.10[illegible]	.0[illegible]		72	[illegible]02	.7[illegible]5
.16	.016	[illegible].70	.0[illegible]	.01[illegible]	1[illegible]	72	2[illegible]	[illegible]
.17	.167	[illegible]	.[illegible]68[illegible]	.0[illegible]		7[illegible]	1[illegible]	.2[illegible]
.26	.2[illegible]	4.[illegible]	[illegible]	.0[illegible]		72	2[illegible]	.[illegible]76
[illegible]	[illegible]	4.[illegible]	[illegible]	.0[illegible]14	141	72	[illegible]	.[illegible]25
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	.[illegible]71
[illegible]	.[illegible]00	[illegible]	[illegible]	[illegible]	1[illegible]	[illegible]	219	[illegible]

SAMPLE 1. BEARING PRESSURE

(Total----------4670 lbs.
)
)Per Sq. In. of
(Projected Area--97 lbs.

Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.09	0.099	5.88	.1025	.0125	150°F	67°	200	.422
.27	.297	5.93	.1034	.0127	156	69	199	.424
.36	.396	5.14	.0894	.0109	152	72	199	.367
.37	.406	5.33	.0929	.0113	155	69	198	.379
.40	.440	5.45	.0950	.0116	149	70	199	.390
.68	.747	5.33	.0929	.0113	157	71	199	.381
.70	.770	5.07	.0883	.0107	151	71	199	.362
2.925	3.220	5.07	.0883	.0107	151	71	199	.362

SAMPLE 1. BEARING PRESSURE

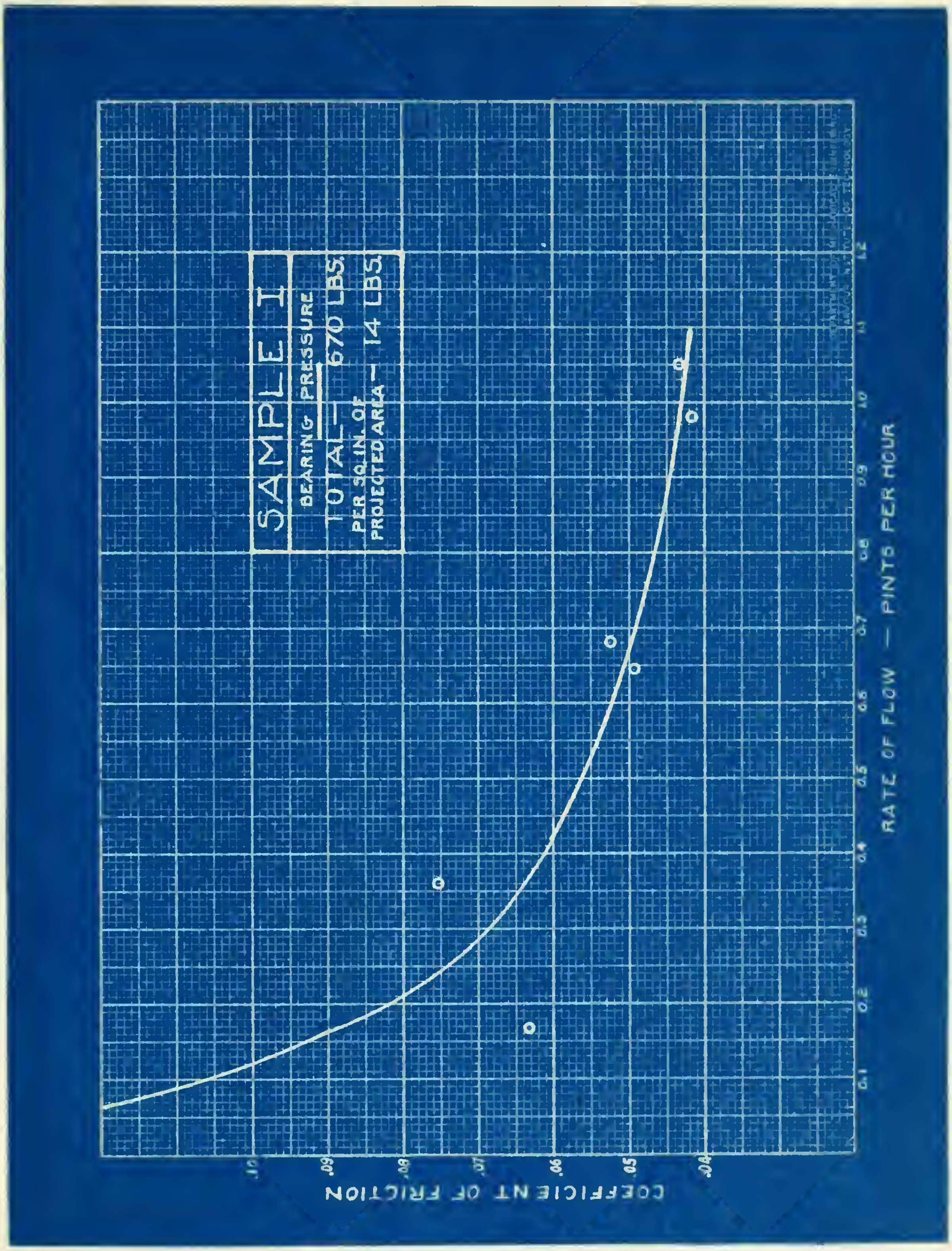
(Total---------6670 lbs.
)Per Sq. In. of
(Projected Area-139 lbs.

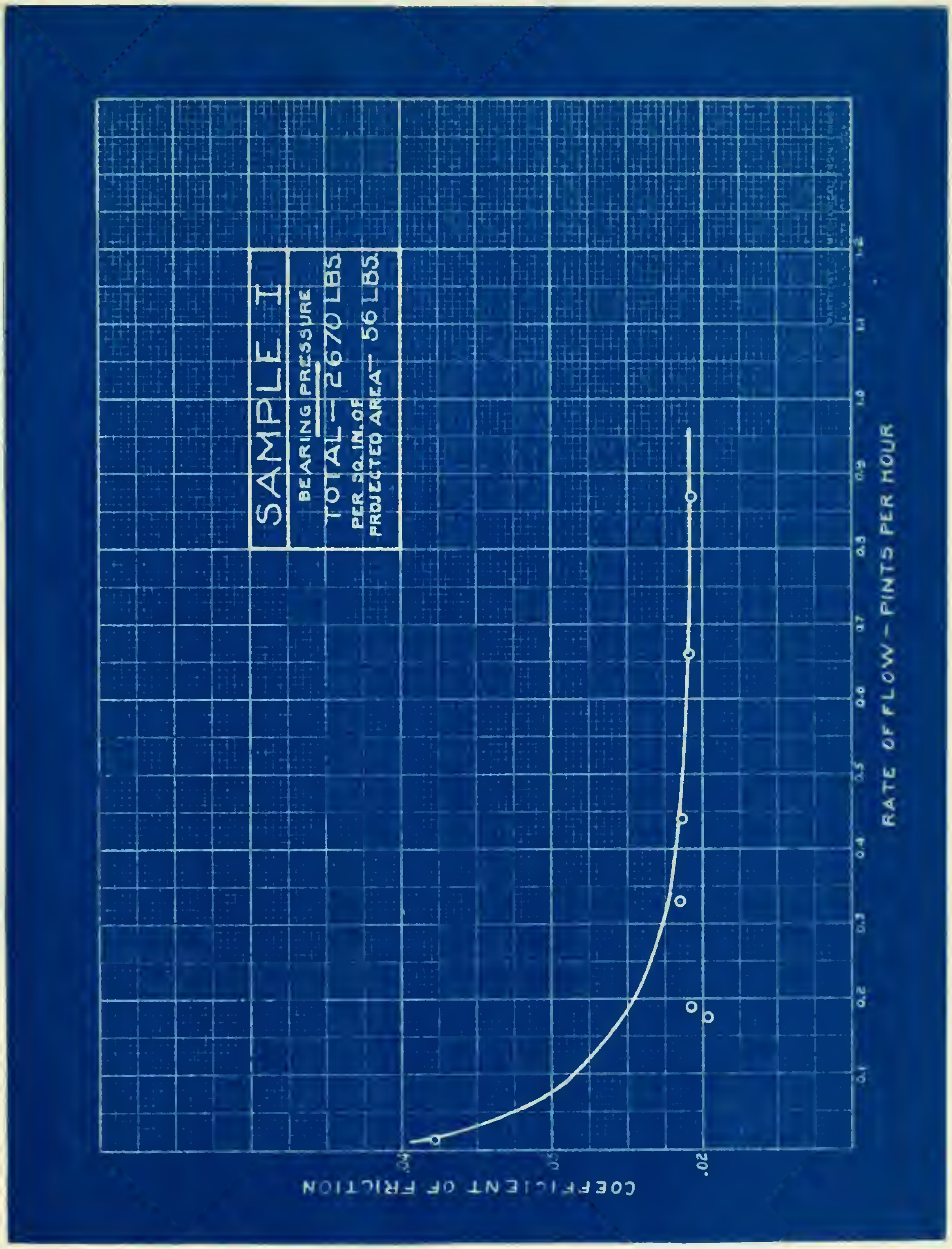
Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.04	0.044	5.77	.1005	.00870	164°F	68°F	199	.412
.24	.264	5.70	.0993	.00860	164	70	197	.403
.32	.352	5.56	.0969	.00839	159	69	198	.395
.34	.374	5.46	.0952	.00825	160	73	199	.391
.54	.594	5.30	.0924	.00800	159	78	198	.377
1.17	1.285	5.22	.0909	.00786	158	76	197	.369

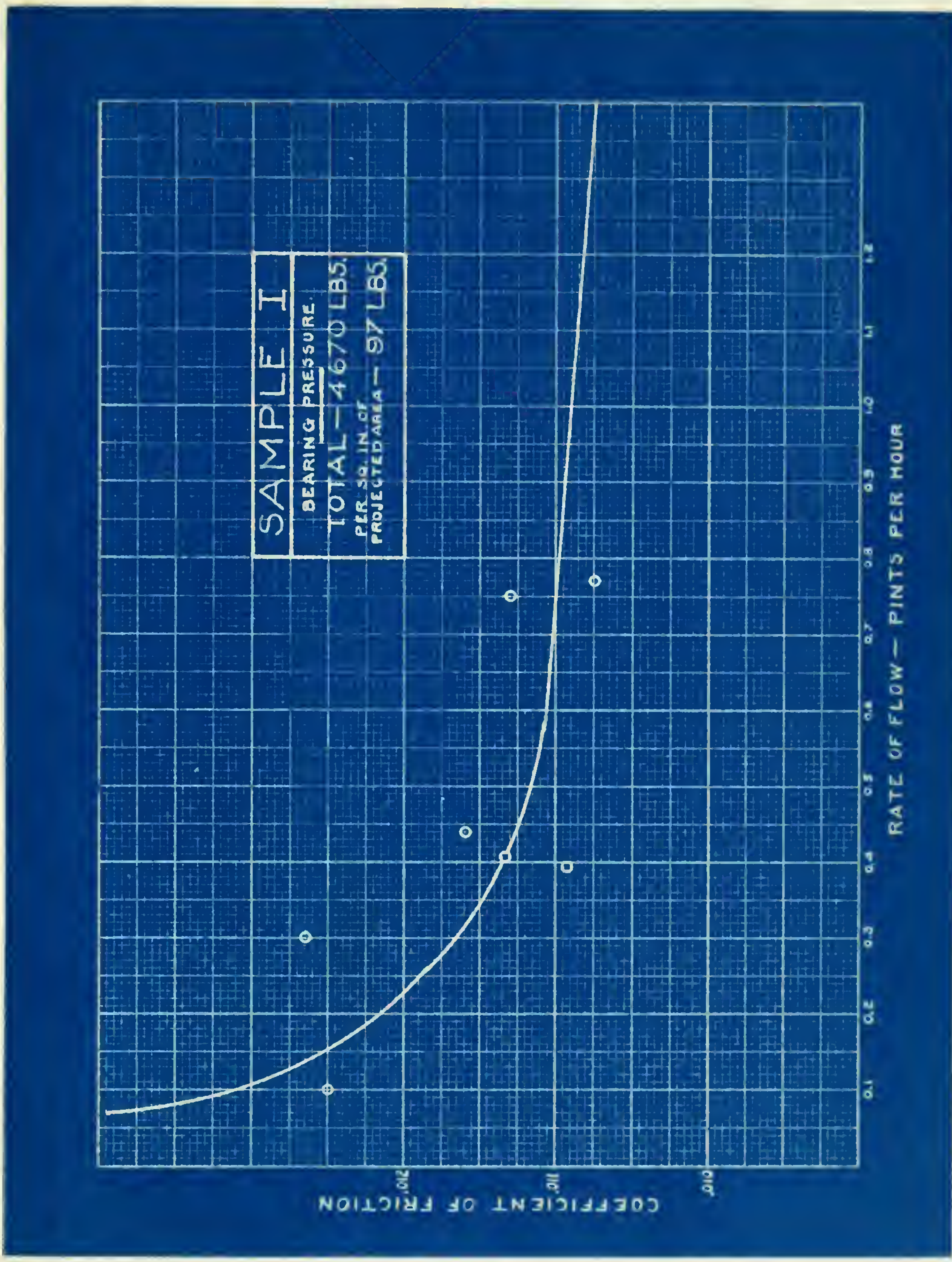
SAMPLE 1. BEARING PRESSURE

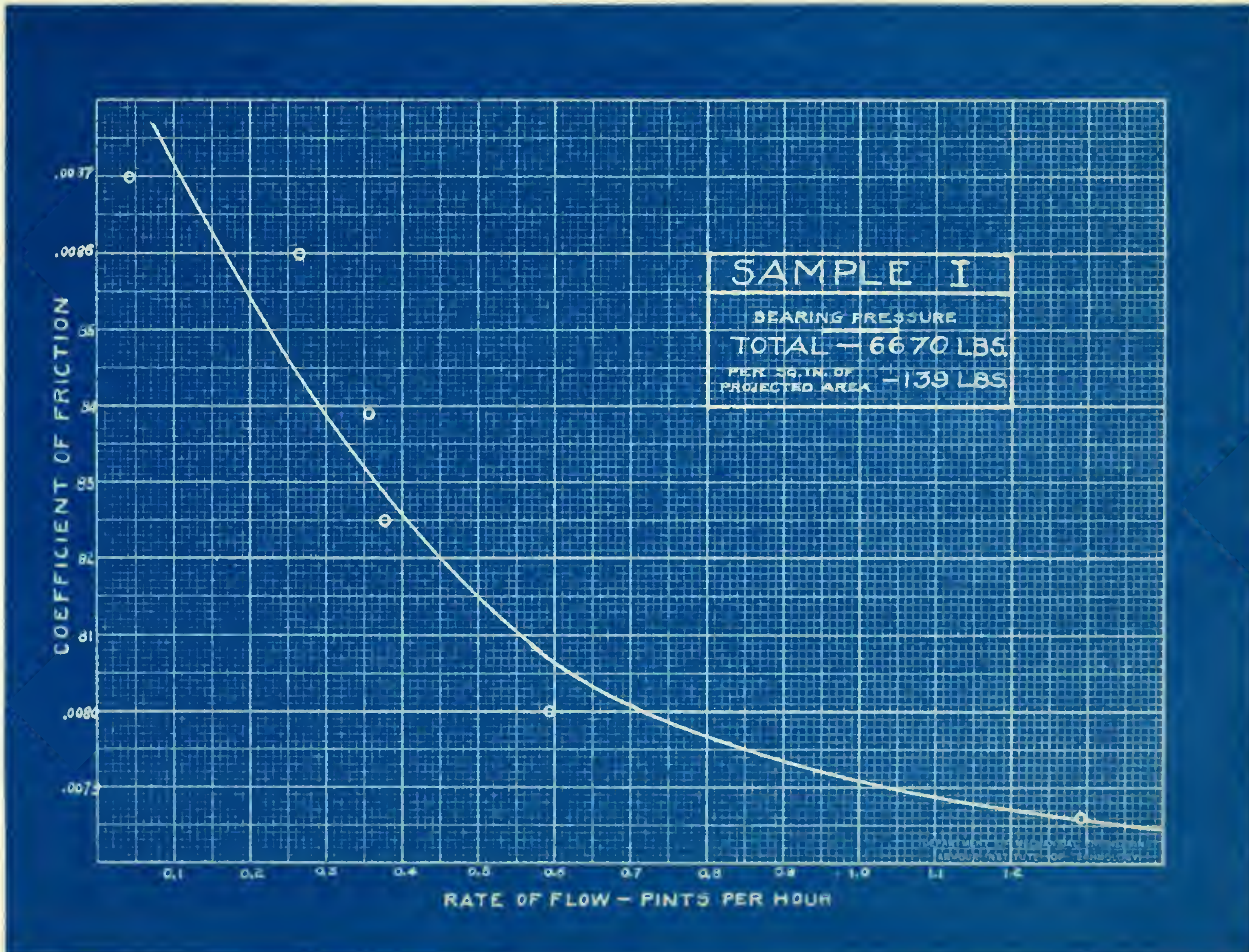
(Total----------8670 lbs.
(Per Sq. In. of
(Projected Area-181 lbs.

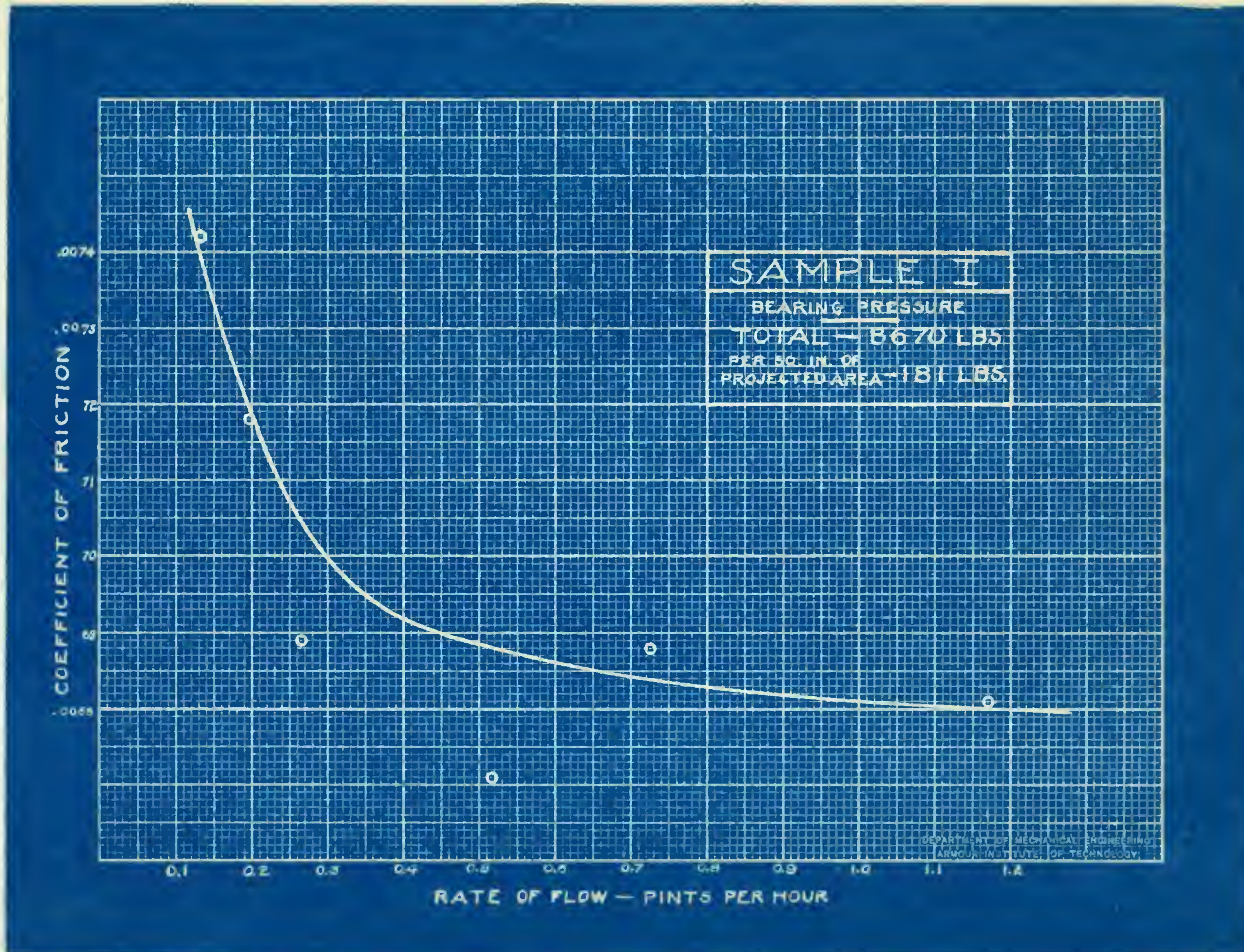
Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.12	0.132	6.40	.1115	.00742	158°F	69°	198	.455
.18	.198	6.20	.1080	.00718	160	72	198	.440
.24	.264	5.96	.1038	.00689	168	78	198	.423
.47	.516	5.80	.1010	.00671	167	79	198	.412
.66	.725	5.95	.1036	.00688	167	70	197	.420
1.07	1.178	5.88	.1025	.00681	169	75	198	.418

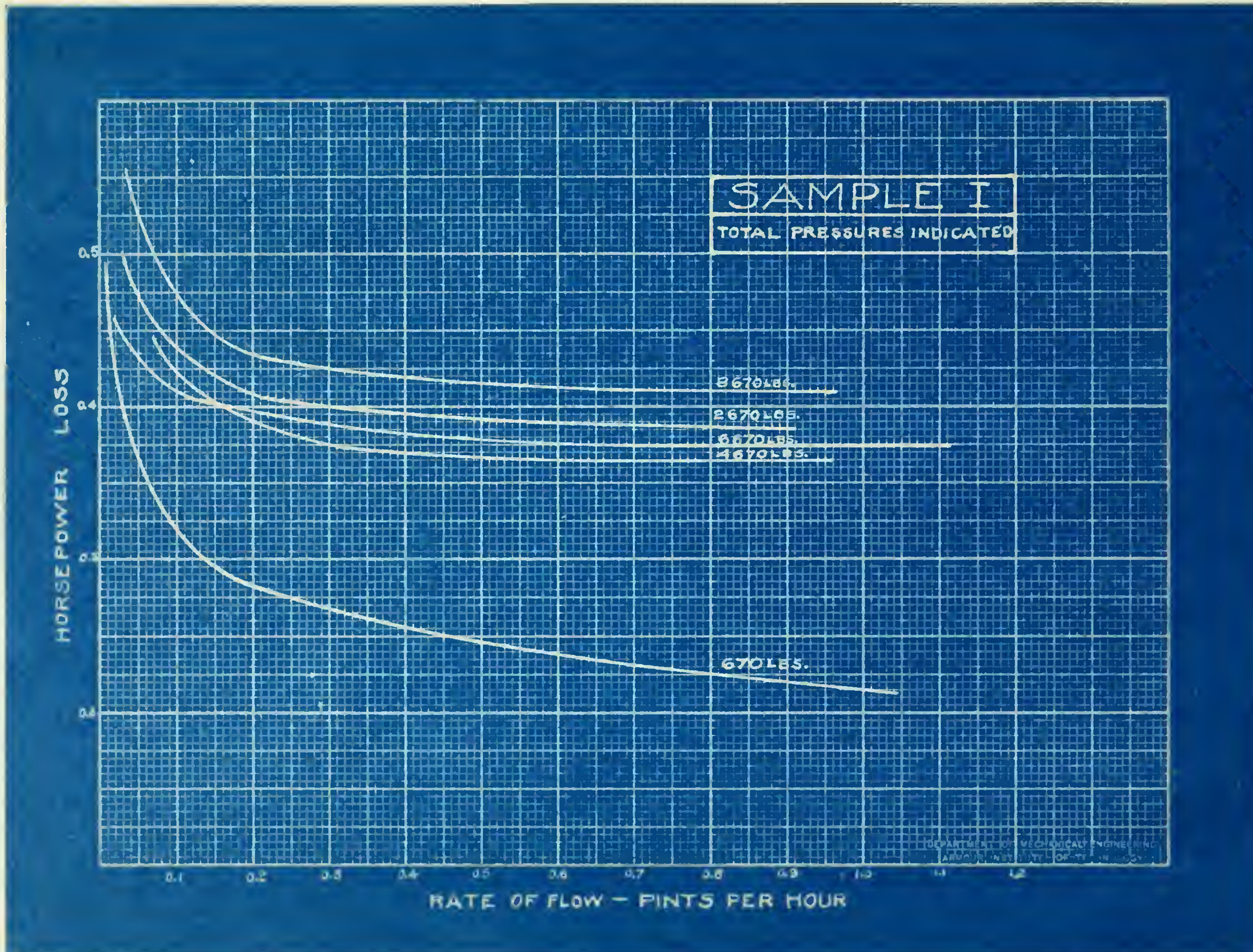

SAMPLE I
BEARING PRESSURE
TOTAL — 670 LBS.
PER SQ.IN. OF PROJECTED AREA — 14 LBS.
RATE OF FLOW — PINTS PER HOUR
COEFFICIENT OF FRICTION
.10
.09
.08
.07
.06
.05
.04
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2


SAMPLE I
BEARING PRESSURE
TOTAL — 2670 LBS.
PER SQ. IN. OF PROJECTED AREA — 56 LBS.
RATE OF FLOW — PINTS PER HOUR
COEFFICIENT OF FRICTION
.04
.03
.02
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2


SAMPLE I
BEARING PRESSURE.
TOTAL — 4670 LBS.
PER SQ. IN. OF PROJECTED AREA — 97 LBS.
RATE OF FLOW — PINTS PER HOUR
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
COEFFICIENT OF FRICTION
.012
.011
.010


SAMPLE I
BEARING PRESSURE
TOTAL — 6670 LBS.
PER SQ. IN. OF PROJECTED AREA —139 LBS.
COEFFICIENT OF FRICTION
RATE OF FLOW — PINTS PER HOUR


SAMPLE I
BEARING PRESSURE
TOTAL — 8670 LBS
PER SQ. IN. OF PROJECTED AREA — 181 LBS.
COEFFICIENT OF FRICTION
.0074
.0073
72
71
70
69
.0068
0.1
0.2
0.3
0.4
0.7
0.8
0.9
1.0
1.1
1.2
RATE OF FLOW — PINTS PER HOUR
DEPARTMENT OF MECHANICAL ENGINEERING
ARMOUR INSTITUTE OF TECHNOLOGY


SAMPLE I
TOTAL PRESSURES INDICATED
8670 LBS.
2670 LBS.
6670 LBS.
4670 LBS.
670 LBS.
HORSEPOWER LOSS
RATE OF FLOW — PINTS PER HOUR

SAMPLE 2

Physical Characteristics.

Color	Blueish Yellow
Baumé (Degrees)	20
Specific Gravity	.9333
Pounds per Gallon	7.78
Flash Point	365°F
Fire "	390°F
Chill "	12°F
Viscosity (referred to distilled water at 150°F)	1.89

SAMPLE 2. BEARING PRESSURE

Total ----------- 670 lbs.
Per Sq. In of Projected Area -- 14 lbs.

Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.01	0.010	2.44	.0425	.0368	107°F	65°	198	.174
.05	.031	2.21	.0387	.0335	109	67	198	.158
.06	.062	1.9	.0332	.0288	113	79	201	.137
.075	.077	1.8	.0314	.0272	109	68	198	.128
.16	.165	1.9	.0332	.0288	113	78	202	.138
.26	.268	1.8	.0314	.0272	108	81	200	.122
.41	.422	1.58	.0277	.0240	106	73	204	.116
.50	.516	1.5	.0262	.0225	105	72	203	.109
.80	.825	1.3	.0228	.0198	104	71	202	.095
.96	.990	1.3	.0228	.0198	103	70	204	.095

SAMPLE 2. BEARING PRESSURE

(Total---------2670 lbs.
)
)Per Sq. In. of
(Projected Area--56 lbs.

Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.04	0.041	4.51	.0786	.01[illegible]	109°F	70°	200	.324
.11	.113	3.79	.0658	.0144	117	69	199	.270
.19	.196	3.43	.0599	.0123	120	68	201	.248
.29	.299	3.10	.0546	.0117	121	73	200	.223
.42	.433	3.05	.0532	.0115	119	69	193	.212
.66	.681	2.90	.0506	.0109	121	74	200	.208
.72	.742	2.88	.0503	.0109	121	74	200	.207
1.42	1.465	3.00	.0523	.0113	119	81	200	.215

S A M P L E 2. BEARING PRESSURE (Total---------4670 lbs.
)
)Per Sq. In. of
(Projected Area--97 lbs.

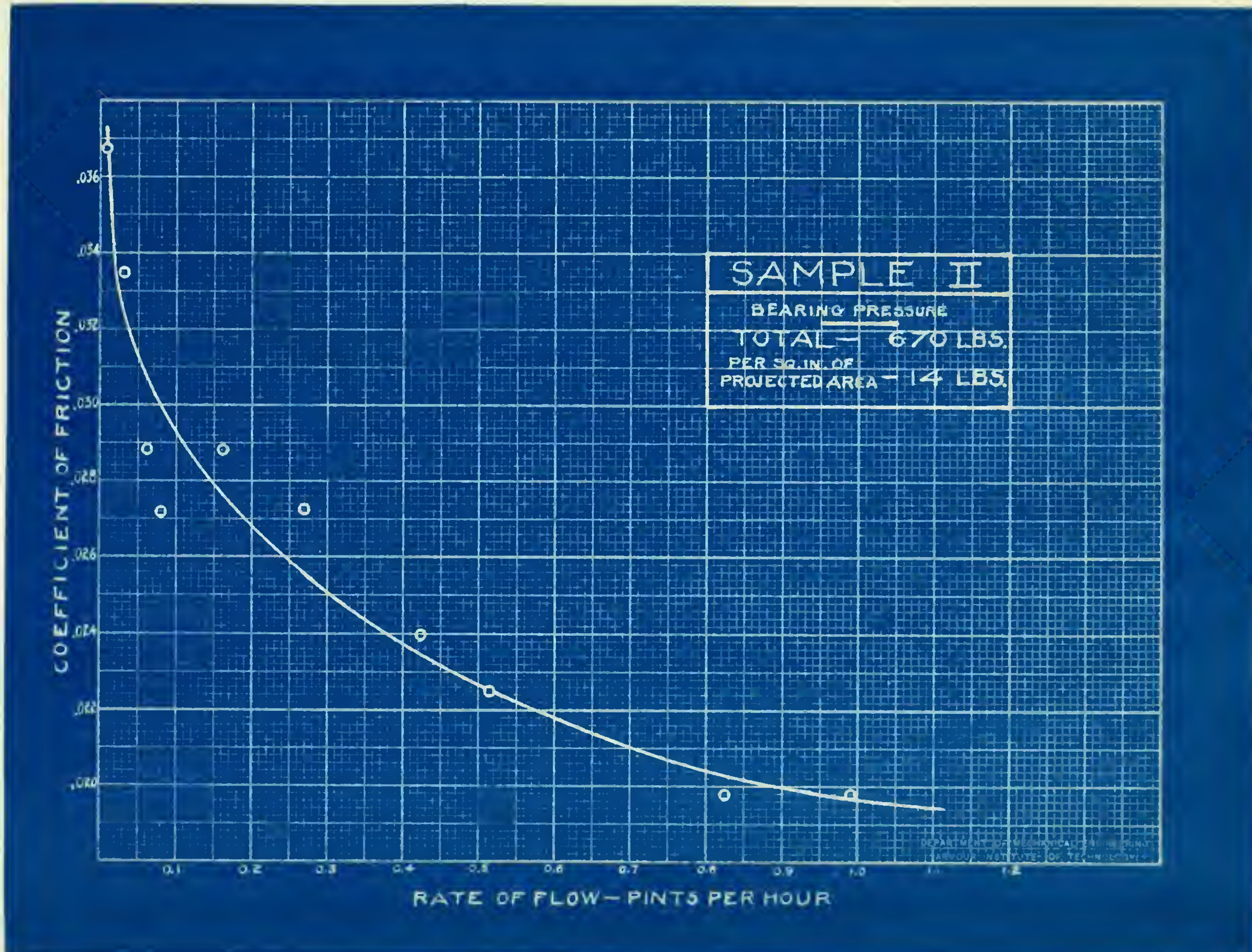
Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.01	0.010	5.95	.1037	.01287	153°F	78°	199	.455
.07	.072	4.39	.0765	.00948	147	71	198	.312
.10	.103	4.17	.0727	.00901	139	73	197	.275
.16	.166	4.00	.0698	.00866	139	71	198	.285
.42	.433	3.82	.0666	.00826	137	72	198	.272
.90	.928	3.70	.0645	.00800	136	72	199	.264
1.28	1.320	3.68	.0645	.00800	135	72	198	.262

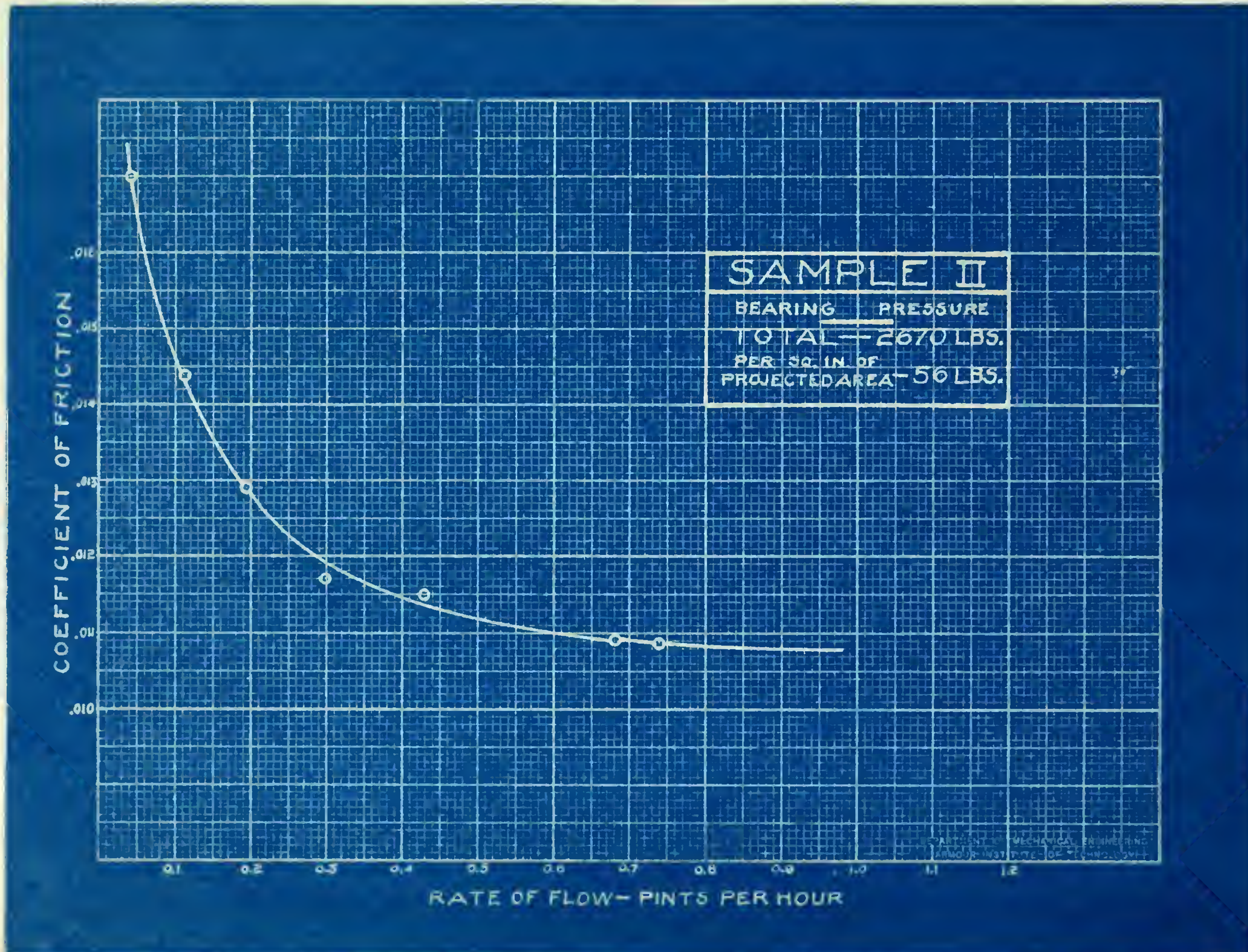
S A M P L E 2. BEARING PRESSURE (Total----------6670 lbs.
)Per Sq. In. of
(Projected Area-139 lbs.

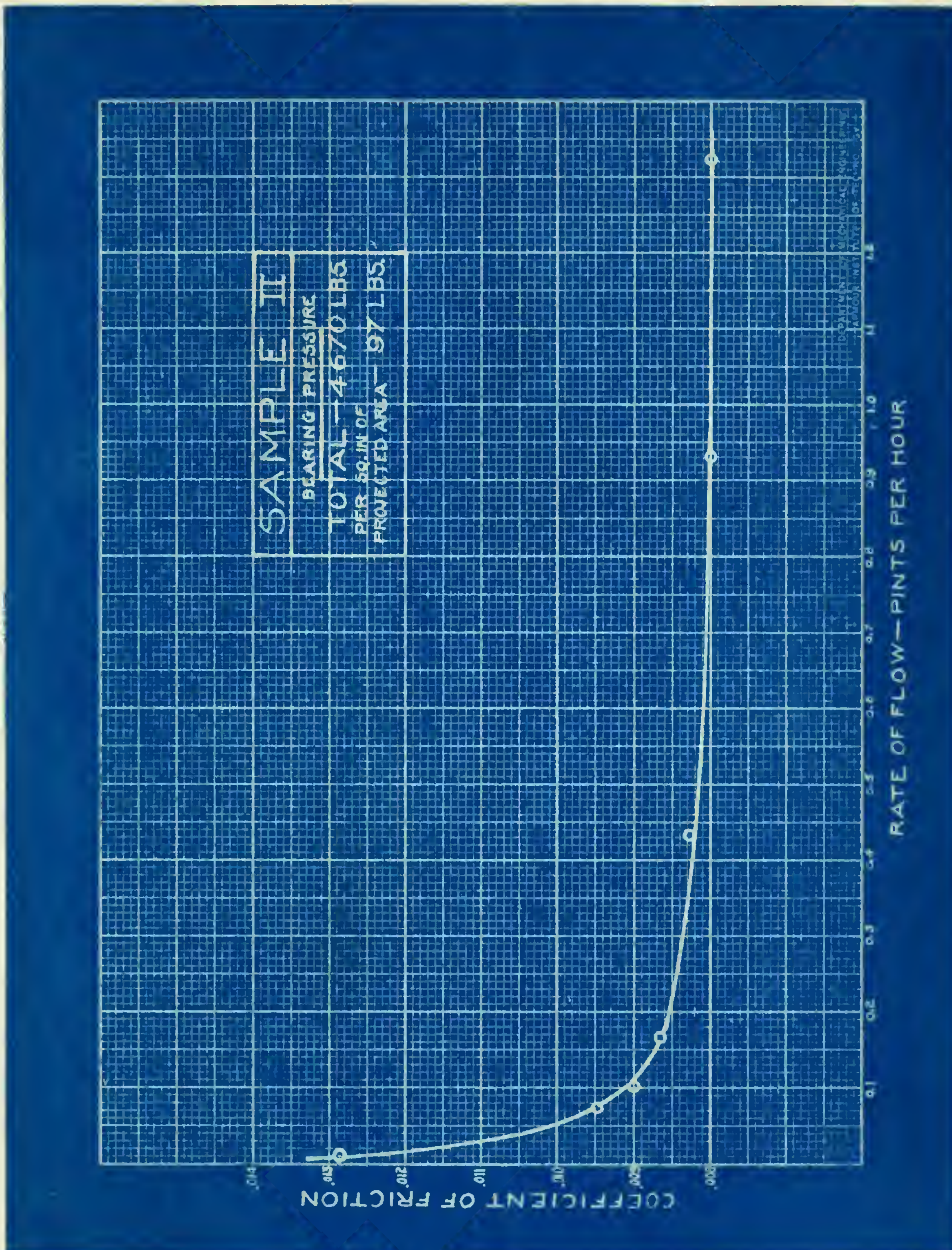
Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.02	0.021	6.40	.1115	.00966	159°F	78°	199	.460
.04	.041	5.70	.0993	.00860	145	67	198	.405
.06	.062	5.22	.0909	.00787	160	77	198	.371
.11	.115	5.00	.0872	.00750	146	80	203	.364
.165	.170	4.96	.0864	.00747	151	74	198	.352
.17	.175	4.87	.0849	.00735	146	67	198	.346
.46	.475	4.70	.0819	.00709	143	66	201	.341
1.02	1.051	4.74	.0825	.00714	146	71	199	.338

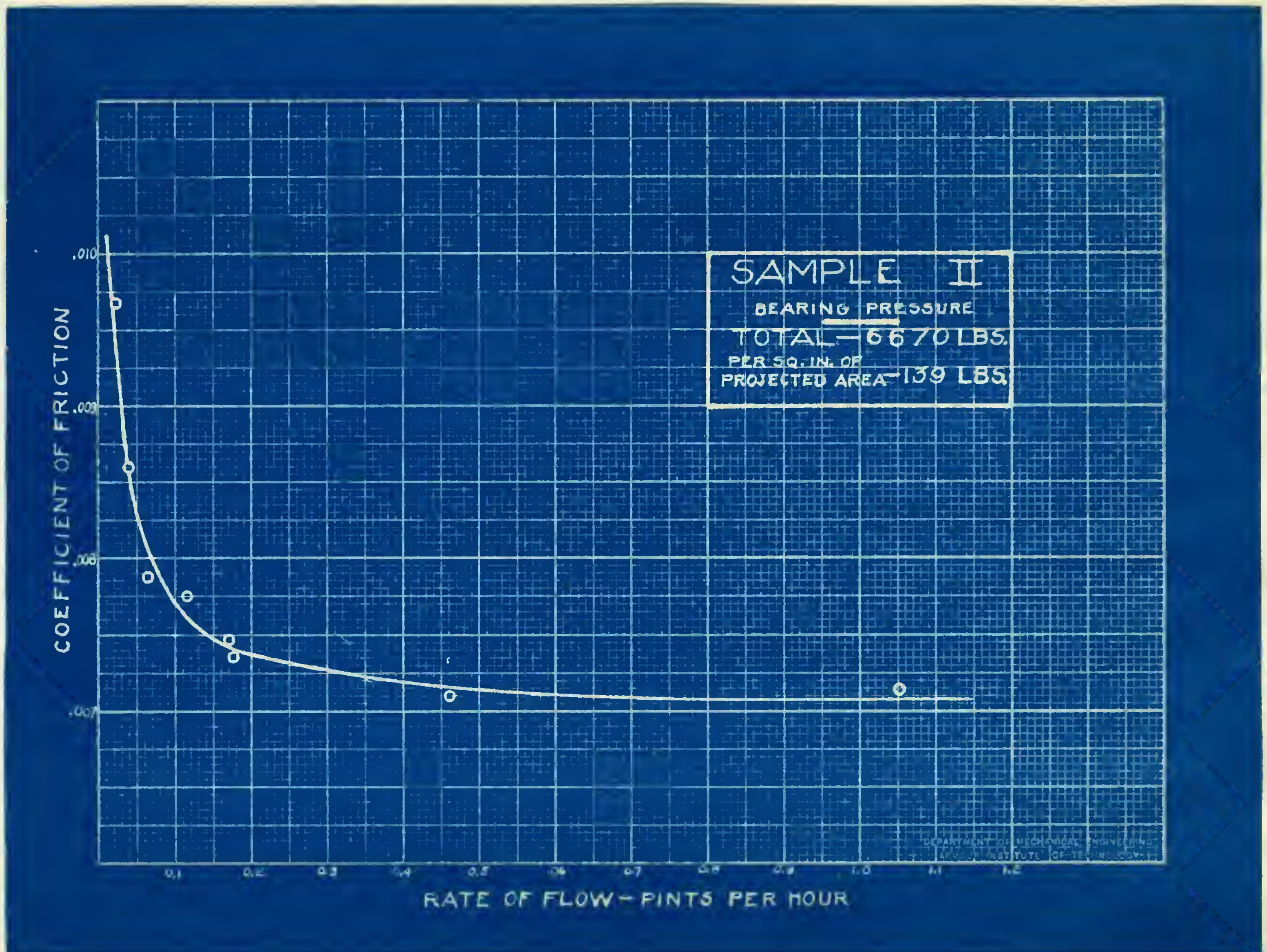
S A M P L E 2. BEARING PRESSURE (Total----------8670 lbs.
(Per Sq. In. of
(Projected Area-181 lbs.

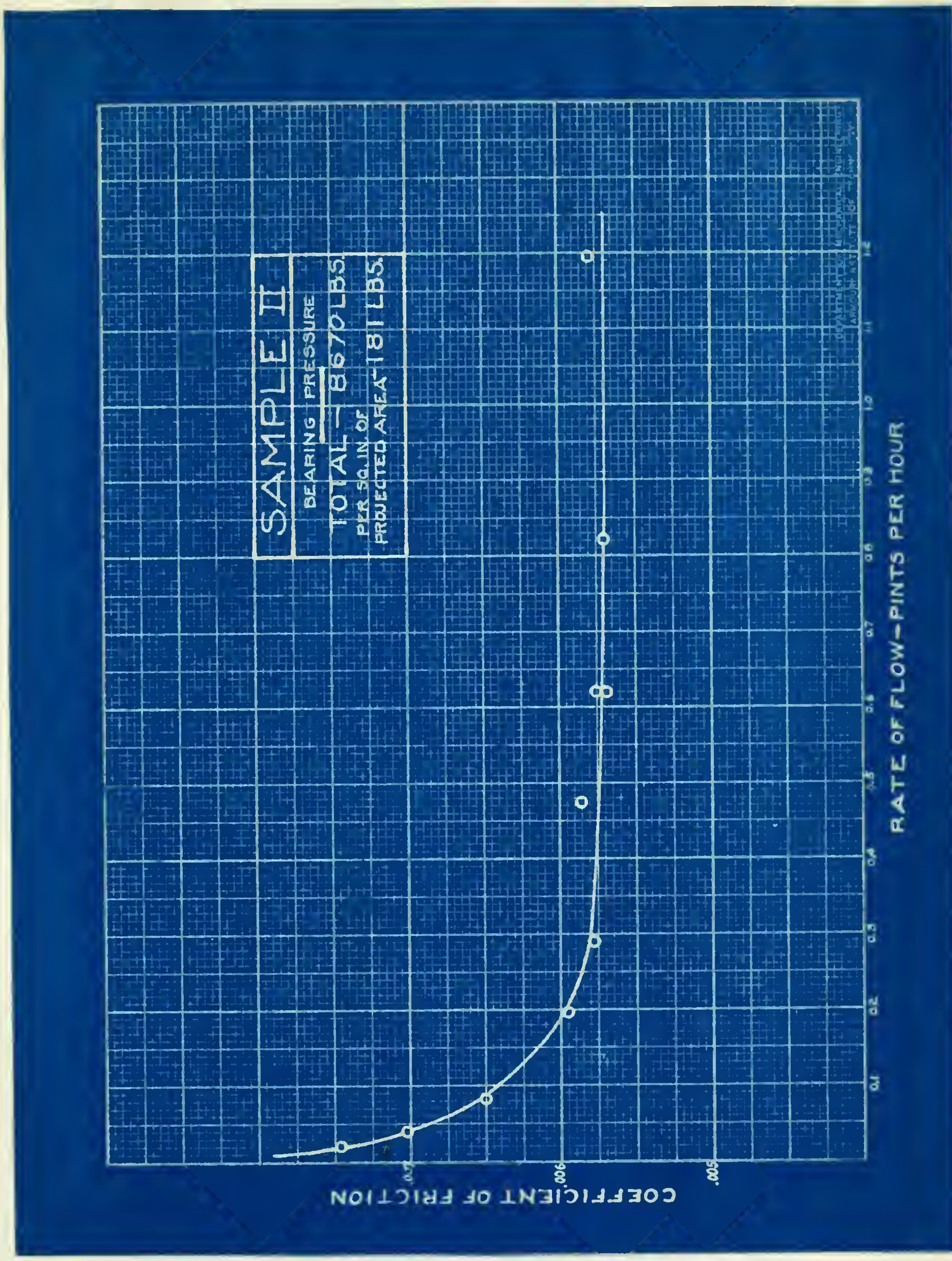
Rate of Flow		Deflection	Sine	Coef. of	Bearing	Room	R.P.M.	H.P.
Lbs/Hr	Pts/Hr	(Degrees)	of Angle	Friction	Temp.	Temp		Loss
0.02	0.021	6.43	.1120	.00745	169°F	74°	198	.455
.04	.041	6.07	.1057	.00702	166	75	198	.431
.08	.082	5.61	.0977	.00650	160	71	201	.405
.19	.196	5.14	.0896	.00596	158	72	202	.373
.28	.289	4.97	.0866	.00576	151	68	198	.353
.46	.474	5.06	.0882	.00586	157	71	203	.370
.60	.619	4.90	.0854	.00567	152	71	198	.348
.60	.619	4.95	.0863	.00574	156	71	202	.359
.78	.805	4.93	.0858	.00571	155	71	202	.357
1.16	1.196	5.0	.0872	.00580	147	69	198	.355

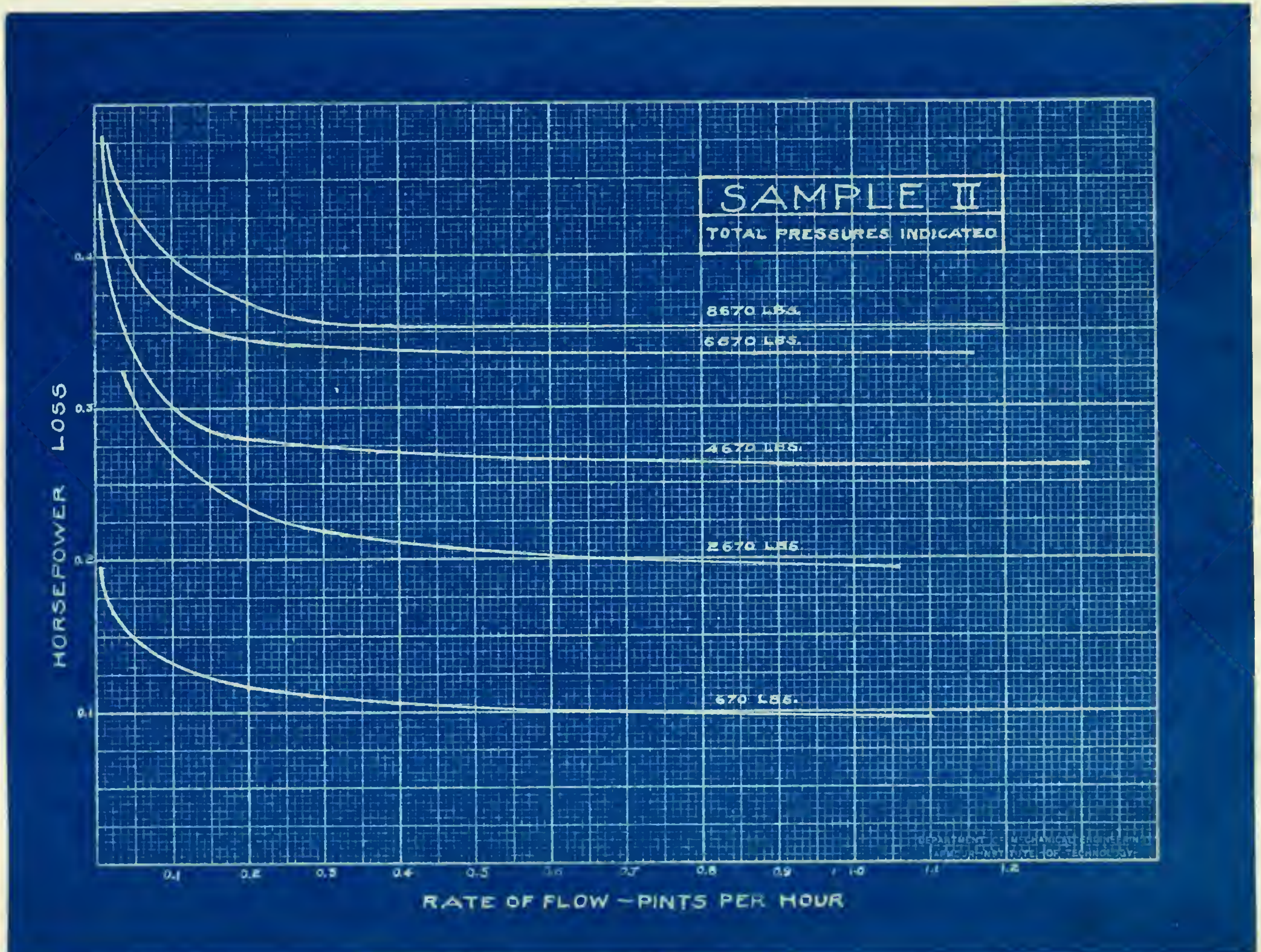

SAMPLE II
BEARING PRESSURE
TOTAL — 670 LBS.
PER SQ. IN. OF PROJECTED AREA — 14 LBS.
COEFFICIENT OF FRICTION
.036
.034
.032
.030
.028
.026
.024
.022
.020
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
RATE OF FLOW — PINTS PER HOUR


SAMPLE II
BEARING PRESSURE
TOTAL — 2670 LBS.
PER SQ. IN. OF PROJECTED AREA — 56 LBS.
COEFFICIENT OF FRICTION
.016
.015
.014
.013
.012
.011
.010
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
RATE OF FLOW — PINTS PER HOUR


SAMPLE II
BEARING PRESSURE
TOTAL—4670 LBS.
PER SQ. IN. OF PROJECTED AREA—97 LBS.
RATE OF FLOW—PINTS PER HOUR
COEFFICIENT OF FRICTION


SAMPLE II
BEARING PRESSURE
TOTAL—6670 LBS.
PER SQ. IN. OF PROJECTED AREA—139 LBS.
COEFFICIENT OF FRICTION
.010
.009
.008
.007
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
RATE OF FLOW—PINTS PER HOUR
DEPARTMENT OF MECHANICAL ENGINEERING


SAMPLE II
BEARING PRESSURE
TOTAL — 8670 LBS.
PER SQ. IN. OF PROJECTED AREA 181 LBS.
RATE OF FLOW—PINTS PER HOUR
COEFFICIENT OF FRICTION
.007
.006
.005
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2


SAMPLE II
TOTAL PRESSURES INDICATED
8670 LBS.
6670 LBS.
4670 LBS.
2670 LBS.
670 LBS.
HORSEPOWER LOSS
0.4
0.3
0.2
0.1
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
RATE OF FLOW — PINTS PER HOUR
DEPARTMENT OF MECHANICAL ENGINEERING
ARMOUR INSTITUTE OF TECHNOLOGY

CALCULATIONS FOR

DETERMINATION OF MOMENT.

-------o------

Determination No.	1	2	3
Distance between Supports (inches) --	27.12	-31.12	-34.31
Weight of Scales (lbs)--------------	85.2	-75.3	-69.7
Weight of Strut (lbs) --------------	16.5	-16.5	-16.5
Net Weight (lbs) -------------------	68.7	-58.8	-53.2
Moment (inch lbs) ------------------	1861	1831	1826

Average Moment = 1840 in. lbs.

Constant of Machine = $\frac{1840}{2.25}$ = 820. where 2-1/4" is the radius of the journal.

This moment entered as a factor only in runs on Sample 1 for total pressures of 670 lbs. and 2670 lbs. For all other runs the moment was smaller due to a faising of the ball, and necessitated calculation as follows:

Weight of Ball-----------------------177.5 lbs.

Distance moved---------------------3-1/32 inches

Decrease in Moment----177.5(3.03) = 540in. lbs.

New Moment------------1840 – 540 = 1300 in.lbs.

New Constant of Machine----------- $\frac{1300}{2.25}$ = 577

SAMPLE CALCULATIONS

Oil Sample 1.

Total Pressure 2670 lbs.---------Run No.1.

Data as obtained:-

Pounds fed per hour ---------- .01

Average Deflection (e) ----- 7.06°

R.P.M. -------------------- 202

Constants:-

Width of brass -------- (b) --- 3 inches

Length of brass ------ (l) --- 8 inches

Constant of Machine -- (C) - 820

Spring Tension ------- (T) 1000 lbs.

Weight of Machine ---- (W) - 670 lbs.

Total projected area (a) --- 2bl = 2(24) = 48 sq.in.

" pressure (P) --- (2T + W) = 2000 + 670 = 2670 (lbs.

It was assumed in all runs that the total pressure was evenly distributed on both bearings, even though the weight of the machine rested wholly on the top bearing.

Then:-

Pressure per sq. in (p) -- $\frac{P}{a} = \frac{2670}{48} = 56$ lbs.

The coefficient of Friction (f) ---- $\frac{C \sin e}{P}$

$$\frac{C \sin \theta}{P} = \frac{820 \sin 7.06}{2670}$$

$$= .0378$$

Horse Power Loss (H.P) -- .00001586NCr sin θ

.00001586(202)820(2.25).1230 = .725

Oil Sample 2.

Total Pressure 670 lbs. Run No. 1.

Data as obtained:-

Pounds fed per hour ------- .01

Average deflection (e) ---- 2.44°

R.P.M. ------------------- 198

Constants:-

Spring Tension ----- (T) -- 0

Weight of Machine -- (W) -- 670 lbs.

Constant of Machine- (C) -- 577

Total Pressure -(P)----- 2T + W = 670 lbs.

Pressure per sq. in. ------ $\frac{P}{a}$ = 14 lbs.

Coefficient of Friction -(f)----- $\frac{C \sin e}{P}$

$$\frac{C \sin e}{P} = \frac{577 \sin 2.44}{670}$$

$$= .0368$$

Horse Power Loss----(H.P)- = .00001586NCr sin e

$$.00001586NCr \sin e = .00001586(198)577(2.25).0425$$

$$= .174$$

In view of the fact that reliable results are obtained only through the use of accurate instruments, great care was teken with the scales and thermometers used, to obtain the proper correction factors. The knife edges of the scales were cleaned so as to render the apparatus more sensitive, and the scales were tested with standard weights, both before and after the completion of the experiment. The thermometers were similarly tested against standard instruments; and all the results herein tabulated are corrected for errors. Effort was made in every direction to make the results reliable.

In testing sample 1, it was found at first that considerable time was consumed in allowing the machine to settle down to a state of constant conditions. Even after such a point had been reached the data as recorded on the running log showed considerable variation, especially at low rates of feed. These discrepancies were probably due to the light body of the oil, which was unable to withstand the pressure when a very slight irregularity in the flow resulted from eddy currents or pockets, and allowed the film on the bearing to "streak". The higher rates of flow overcame this difficulty to a great extent,

since the supply was sufficiently large to make slight variations negligable; but the time element entering into the settling down to constant conditions was not much reduced.

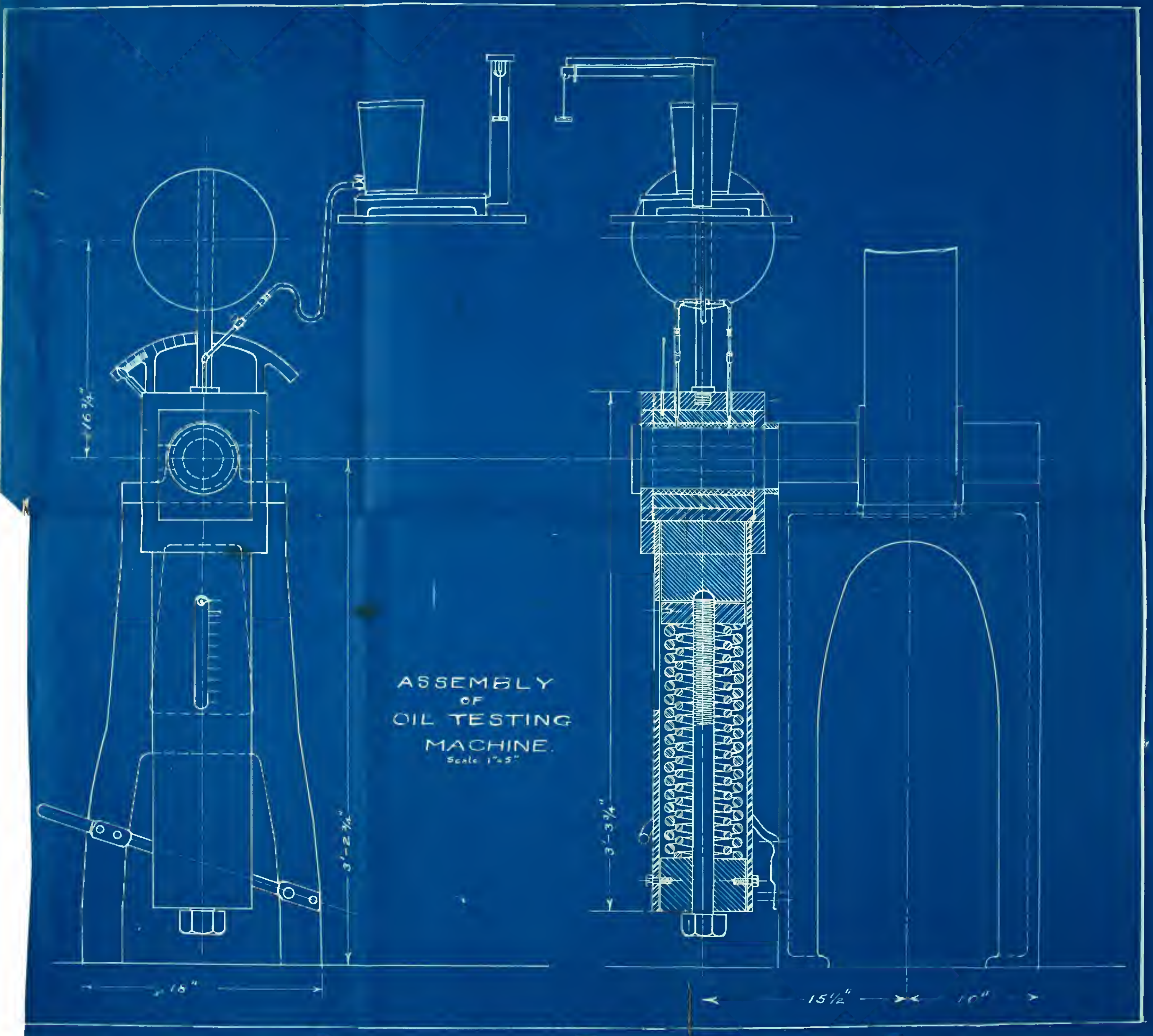
Sample 2, however, which was of heavier body, though slightly lower in viscosity, proved less troublesome in every way; for it not only took less time to bring about constant conditions, but the readings showed practically no variations during the runs. Since it was not until the highest pressure was reached that any irregularities were noticed it was concluded that the heavier body was responsible for this smoothness of running. In addition to the fact that the second specimen of oil showed better running qualities than the first, figures obtained show it to be more economical as regards friction loss. Comparing the runs of the two oils for each pressure, it will be seen that in every case the coefficient of friction for sample 2 is less than that of sample 1. The data on the horse power loss also brings this out very effectively, which can be seen at a glance by noting the curves. It may be noticed on the coefficient curves of Sample 1 that the point where the curve begins to

flatten out is much further out than the corresponding point on the curves for Sample 2. That is, the point beyond which the increase of rate of feed produces no appreciable saving is a much lower value in the case of Sample 2 than in the case of Sample 1 which is another point in favor of the second specimen tested.

The data from Sample 1 proved to be very inconsistent as the plotted points indicate; in almost every case for that specimen the points distributed themselves in a "shot-gun pattern" It was not so for Sample 2; for almost all of the points fell along a very well defined path. Thus, in drawing the curves for the first sample, reference was made to the characteristic shape as found in Sample 2. The horse power curves of Sample 1 show the absurdity pf these results much more clearly, for the horse power loss at the 2670 lb. pressure appears greater than that at 4670 or 6670 lbs. pressure. This is considerably different from the other case in which the curve seems to rise by even steps as the pressure increases. In as much as the pressure on the bearing enters as a factor into the coefficient of friction, a wide range of figures resulted which

which rendered it impractical to superpose the curves as was done in the case of the horse power, thus hiding the freakish nature of the results.

A study of the data and curves herein contained irrespective of comparison of the oils, will show the advantage of the continuous system of lubrication over the restricted feed method. Such a system permits the flooding of the bearing at all times, thus reducing the friction to a minimum, which saving may be traced back to the coal pile, and figured up in dollars and cents. Then, too, the greater part of the oil which is lost in the restricted feed method, may be reclaimed and used over and over again, which is another distinct saving. Of course, small plants are not to be considered in recommending the installation of these flooded systems; but the large stations where the saving amounts to several hundred horse power find this method almost necessary for economical operation.


ASSEMBLY
OF
OIL TESTING
MACHINE.
Scale 1"=5"
16 3/4"
3'-2 3/4"
3'-3 3/4"
18"
15 1/2"